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FEASIBILITY OF A SINGLE COMMON POWERTRAIN LUBRICANT: HYDRAULIC SYSTEM INVESTIGATIONS AT LOW TEMPERATURES

INTERIM REPORT
TFLRF No. 411

by
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for
U.S. Army – TARDEC
Force Projection Technologies
Warren, Michigan

Contract No. DAAE-07-99-C-L053 (WD42)

January 2011

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14. ABSTRACT:

This project was conducted in support of the feasibility study for the U.S. Army Single Common Powertrain Lubricant (SCPL). U.S. Army vehicles were surveyed to determine the types of hydraulic system pumps that are used. The hydraulic system from a representative vehicle (6K) Rough Terrain Fork Lift was selected as the basis for a bench test apparatus that was used to investigate the low temperature flow performance of various fluids.

The test rig was assembled and low temperature tests were conducted. Results indicated that a commercially available dedicated hydraulic fluid had outstanding low temperature flow performance. Given the observed low temperature flow performance of the current fielded Arctic Engine Oil, MIL-PRF-46167D, it is recommended that the proposed SCPL should have equivalent or lower viscosity at low temperatures.

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EXECUTIVE SUMMARY

The U.S. Army Tank Automotive Research Development & Engineering Center (TARDEC) is investigating the feasibility of replacing multiple specifications for multiple grades of powertrain lubricants with a single common lubricant for tactical and combat vehicles. The proposed new Single Common Lubricant (SCPL) would replace the current arctic lubricant MIL-PRF-46167D. The purpose of this investigation was to survey the potential effects of operation of a hydraulic system when using engine oils that have a higher low temperature viscosity than dedicated hydraulic fluid.

General information was first obtained on the effects of high viscosity oil at low temperatures in a hydraulic system. The primary effect was found to be reduced flowrate from the pump at startup due to cavitation in pump suction lines from excessive pressure drop. The pump cavities do not completely fill during the intake cycle, consequently only discharging a reduced amount of fluid. Reduced flowrate can result in unsafe vehicle operation when the hydraulic power steering or hydraulic power assist brake systems do not receive sufficient flow to respond to operator requirements. Failures or deterioration of hydraulic components may also occur if high loads and excessive use occurs with excessively high viscosity fluids.

A survey of construction, tactical, and combat vehicles currently using MIL-PRF-2104 engine oil in the hydraulic system and general fleet density data were reviewed to identify a vehicle that had a hydraulic system with common characteristics found in other types of Army vehicles. A 6,000 pound (6K) Rough Terrain Fork Lift, NSN 3930-01-158-0849 or 6000M, was selected for further study. This vehicle's hydraulic system consists of two pump circuits. One is an open center system with a fixed displacement tandem gear pump. The second system is a closed center load-sensing system with a variable displacement axial-piston pump. Both systems share the same reservoir.

A test rig was developed to duplicate each of these systems, including the reservoir, suction lines, fittings, and pumps. The test rig was assembled and placed in a refrigerated environmental chamber pump. The intent of the test was to simulate a startup event at engine idle speeds and monitor system pressures and the output flow of the pumps for three different fluids to determine their pumpability at various ambient temperatures. The three fluids evaluated included the previous arctic fluid (MIL-PRF-46167C), the current arctic fluid (MIL-PRF-46167D), and a commercially available, dedicated hydraulic fluid, Exxon Mobil Univis HV-26, with a very high viscosity index.

The results of the testing revealed that the effect of high fluid viscosity at cold temperatures is due to several factors. The effects vary with the pump and the system architecture. For the tandem gear pump, there was a reduction in discharge flow for the previous arctic oil as temperatures were lowered below 40 °F. The combined flowrate of both sections of the tandem pumps continued to drop, producing less flow as temperatures were reduced to -30 °F. The current arctic fluid had very similar performance except that the drop in flowrate occurred at temperatures from 5 °F to 10 °F higher than the previous arctic fluid. Between the two gear pump sections, the pressure and flow characteristics were significantly different. For the smaller of the two pumps, most of the pressure drop occurred in the suction line, resulting in cavitation before the fluid entered the pump. For the larger pump, most of the pressure drop occurred within the pump with cavitation originating within the pumping cavity. For the Mobil Univis fluid, the pump began reducing flowrate at temperatures of approximately 40 °F lower than the previous arctic fluids.

The axial piston variable displacement pump exhibited different cold weather characteristics. For the previous arctic fluid, the flowrate began to reduce at temperatures below 0 °F. For the current arctic fluid the flowrate began to reduce at temperatures below 5 °F. For the Mobil Univis fluid, there was no reduction in flowrate at temperatures as low as -30 °F. The reduction in flow appeared to be due primarily to cavitation internal to the pump because at cold temperatures, once flow rate began to reduce, the inlet pressures at the pump inlet port remained relatively constant.

The current arctic oil has a slightly higher kinematic viscosity than the previous arctic oil and performed marginally poorer in the pump evaluations. The difference in pumping characteristics between current and previous arctic oils is approximately 5 to 10 ºF depending on the pump and its inlet characteristics.

As temperatures are lowered at the idle test speeds, the flow rate for both the current and previous arctic oils begins to reduce, somewhat linearly with temperature. For the gear pump, the flow rate continues to drop over a range of approximately 60 °F until there is essentially no flow. For the variable displacement pump, the flow rate began to start dropping at a much lower temperature, however, once it began dropping, the flow rate dropped much faster, approaching zero flow over a range of approximately 30 °F.

It is difficult to extrapolate this data to actual performance of a hydraulic oil in a vehicle, and much more difficult to extrapolate performance of a given oil in the military fleet. This is due to the wide range of operational differences between vehicles and the wide range of pump inlet geometries, inlet screen restriction, fitting geometry, hose diameter, and hose length present within different classes of vehicles within the fleet.

Overall, given the observed low temperature flow performance of the current arctic engine oil, MIL-PRF-46167D, it is recommended that the proposed SCPL should have equivalent or lower viscosity at low temperatures.

FOREWORD/ACKNOWLEDGMENTS

The U.S. Army TARDEC Fuel and Lubricants Research Facility (TFLRF) located at Southwest Research Institute (SwRI), San Antonio, Texas, performed this work during the period of October 2006 through January 2011 under Contract No. DAAE-07-99-C-L053. The U.S. Army Tank-Automotive RD&E Center, Force Projection Technologies, Warren, Michigan administered the project. Mr. Luis Villahermosa (AMSTA-RBFF) served as the TARDEC contracting officer's technical representative.

The authors would also like to recognize the contribution of the SwRI Engine, Emissions and Vehicle Research Division. Finally, thanks to the administrative and report-processing support provided by Dianna Barrera.

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ACRONYMS AND ABBREVIATIONS

ASTM	American Society for Testing and Materials
°C	Degrees centigrade
cSt	Centistokes
cu in	Cubic Inch
°F	Degrees Fahrenheit
FMTV	Family of Medium Tactical Vehicles
GPM	Gallons per minute
HEMTT	Heavy Expanded Mobility Tactical Truck
Hg	Mercury
HMMWV	High Mobility Multipurpose Wheeled Vehicle
HP	Horse power
lb-in	Pound-inch
LPM	Liters per minute
NSN	National Stock Number
OE/HDO	Oil Engine / Heavy Duty Oil
PLS	Palletized Load System
psi	pounds per square inch
psia	pounds per square inch, absolute
psig	pounds per square inch, gauge
Rev	Revolution
RPM	Revolutions per minute
SCPL	Single Common Powertrain Lubricant
SwRI	Southwest Research Institute
TARDEC	Tank-Automotive RD&E Center
TFLRF	U.S. Army TARDEC Fuels and Lubricants Research Facility
TM	Technical Manual

1.0 OBJECTIVE AND BACKGROUND

The overall objective of the program is to determine the technical and economic feasibility of developing and implementing a Single Common Powertrain Lubricant (SCPL) for use in all combat and tactical equipment, currently using MIL-PRF-2104 and MIL-PRF-46167D lubricant products. To ensure the SCPL has the broadest possible operating range, it is important to understand the effect that lubricants have on the performance of hydraulic system components. One concern is that the viscosity of the proposed SCPL could be too high to flow properly in hydraulic systems operated in cold climates. The previous MIL-PRF-46167C lubricant has a specified maximum kinematic viscosity of 15,000 cSt at -40 °F as compared to the current specification which has a maximum viscosity of 18,000 cSt at -40 °F. The higher viscosity of the current lubricant is a concern. The work reported herein addresses this concern by simulating a military hydraulic system in a cold environment and testing oils to provide relative rankings of their pumping characteristics.

Since it would be a difficult task to test and evaluate a new lubricant in all potential vehicles, as a first step, it was decided to select a vehicle to evaluate that had relatively high use and that had a hydraulic system that was typical of most Army vehicles that use MIL-PRF-2104 or MIL-PRF-46167 as a hydraulic fluid. Once selected, the evaluation of the vehicle hydraulic system could be performed by either testing the complete vehicle in an environmental chamber or by duplicating the hydraulic circuit most affected by cold temperatures and testing it in an environmental chamber. The second option was selected because of the limited availability of the vehicles and an appropriately sized environmental chamber. In addition, a laboratory test rig would more easily permit the installation of additional instrumentation allowing for a more complete analysis of operation in cold environments. The approach taken for this study is as follows:

1. Obtain general information on the high viscosity effects of lubricants operating in cold temperatures on typical hydraulic systems.
2. Survey Army Tactical vehicles for those that currently use MIL-PRF-2104 and MIL-PRF-46167D as a hydraulic fluid.
3. Select high density vehicles of various types and determine the types of hydraulic systems used on them.

4. Select one vehicle as a target vehicle that has a hydraulic system typical of most Army tactical and construction vehicles.
5. Fabricate a laboratory test setup which uses critical components from the target vehicle.
6. Conduct low temperature laboratory tests of various lubricants using the laboratory test setup.
7. Evaluate the test results.

Background information was obtained on the typical effects of operating hydraulic systems in excessively cold environments so that a laboratory test program could be formulated to evaluate the different lubricants in cold environments.

In this report, the terms “lubricant”, “oil”, and “fluid” will be used interchangeably.

2.0 LUBRICANT VISCOSITY EFFECTS AT LOW TEMPERATURES ON HYDRAULIC SYSTEMS

There are many components to a hydraulic system whose performance is affected by high viscosity. In general, the pressure drop of flow through any type of restriction is directly proportional to the flow rate and the dynamic viscosity of the lubricant. (This assumes that the flow condition is laminar, where the Reynolds Number is less than 2000). This is particularly true for the lines and the fittings and passages where fluid enters and exits a component. High pressure drop through lines and components may be tolerated by some components for a short period of time, however, performance may suffer and failure of the components may be at risk. The following is a list of key components and a discussion of the potential effects of high viscosity on their operation.

2.1 HYDRAULIC PUMPS

Hydraulic pumps are perhaps the most susceptible component to be affected by variation in lubricant viscosity and especially high viscosity lubricants. The typical maximum intermittent viscosity recommended for pumps ranges from 1600 cSt to 2160 cSt. The recommended operating range can be from 10 to 43 cSt. The recommended operating range provides an optimum level of the following effects.

Low internal leakage—Leakage increases as viscosity decreases.

Low friction and churning losses—Friction reduces as viscosity decreases.

Good lubrication—Very high viscosity may starve mechanical parts of lubrication and very low viscosity may result in breakdown of fluid film and increase wear.

The maximum viscosity specification is intended to avoid insufficient supply of fluid to the pump or cavitation. When the pressure drop from the reservoir through the suction line and into the interior pumping cavities reaches the point where the dissolved gases in the fluid or the vapor pressure of the fluid is reached, cavitation will occur. The pumping cavities will not completely fill with fluid on the intake stroke leaving a void and then on the discharge stroke the flow out will be limited. The most obvious result is that there will be sluggish operation of the hydraulic functions. A second observation may be that the fluid may become foaming or aerated because of dissolved gas being pulled out of solution. Aerated fluid will result in soft or sloppy operation of hydraulic functions. Aerated fluid may also result in improper operation of control valves. Usually after a warm-up period, when there is no more cavitation, the aerated fluid will clear up and normal operation can resume. Pump manufacturers will commonly specify a minimum suction pressure, typically at 5 inches Hg vacuum (12.2 psia) for normal operation to ensure complete filling and no cavitation.

In some cases operation in cavitating conditions may result in pump failure. A potential failure mode on piston pumps may be that the piston hold-down mechanism may fail, or the piston may become separated from the slipper/shoe. Long term operation with cavitation, or with aerated fluid can result in cavitation erosion "implodes" with a very high concentration of energy. If the imploding bubble is attached to or very close to the surface of critical pump parts, the impact of the imploding bubble can overstress and fatigue the metal, eventually creating pits on the surface. Cavitation erosion is most likely to occur in the valve plate area of pumps, where the erosion can eventually cause additional wear and internal leakage, deteriorating the pump's performance.

Another potential failure mode of pumps in a cold environment with high viscosity fluid occurs when high pressure is too rapidly applied to a pump or when a valve is actuated that causes a variable displacement pump to quickly change displacement, resulting in a momentary high flow rate into the pump case. With high viscosity fluid, an excessive amount of backpressure may occur from the case drain line, back to the reservoir, resulting in over-pressurization of the case. Most pump cases are designed to withstand 50 to 100 psi case pressure. When over-pressurized, one of several failures may occur: the shaft seal may fail; case seals may fail; the case may crack.

Yet another potential pump failure that may occur with high viscosity fluid is due to a lack of lubrication, which usually occurs when high pressure is applied. High viscosity fluid may not be able to adequately reach highly loaded bearing surfaces, such as journal bearing, vane pump vanes, or piston pump slipper/shoes. Again, this failure is most likely to occur when high pressure is applied before the system is properly warmed up.

Startup operation of the pump in low temperature conditions should be at low speeds and low pressures to mitigate the potential damage to the pump until the fluid is warmed up.

2.2 CONTROL VALVES

Directional control valves and other types of pressure control and flow control valves may not function properly with high viscosity fluid. Valves may be slow to respond or may not respond at all. Orifices, which normally are not very sensitive to normal temperature variations when the flow rate is turbulent, become very sensitive at high viscosities.

Some control valves have low limits on the backpressure of lines that return fluid back to the tank. Excessive back pressure from high viscosity fluid flow can cause failure of low pressure portions of a valve.

2.3 ACTUATORS

Most actuators, such as hydraulic cylinders do not exhibit problems with high viscosity fluids. Typically problems with pumps and valves supersede cylinder problems. Hydraulic motors, on the other hand can have failure problems. Hydraulic motor problems are most likely to be excessive case pressure or seal pressure due to high backpressure on the return line, causing a failed shaft seal or cracking the case as in a pump.

2.4 FILTERS AND COOLERS

Most filters and coolers are designed for a maximum gage pressure and a maximum pressure drop. Excessive pressure drop through a filter can cause the filter to collapse and form a hole, reducing its filtering capability. Most filters are equipped with a bypass relief valve that allows the fluid to bypass the filter media when a certain level of differential pressure occurs. The filter housing has to contain the gage pressure, which is a combination of the differential pressure and the downstream backpressure. Failure of the filter housing or seals can occur if the rated pressure is exceeded.

Coolers are affected by high viscosity fluids similarly to filters. Coolers often have bypass valves to minimize the effect of high viscosity fluids.

3.0 SURVEY OF ARMY TACTICAL AND CONSTRUCTION VEHICLES AND HYDRAULIC SYSTEMS

3.1 VEHICLE SURVEY

Army Program Manager Offices were contacted to assess the usage of MIL-PRF-2104 lubricant as a hydraulic fluid. A brief summary of the findings can be found in Table 1.

Table 1. Summary of Engine Oil Usage in Army Tactical and Construction Vehicles

Equipment Category	Total Number of Models	Number of Models Using Engine Oil for Hydraulic System
Construction Equipment	69	30
Material Handling	16	13
Bridging Systems	11	5

Other vehicles also known to use engine oil in some hydraulic fluid applications:

- HEMTT
- PLS
- M88 Recovery Vehicle
- HMMWV
- FMTV Recovery Vehicle

Technical Manuals were reviewed for selected high density vehicle models out of each category. The TM's provided basic information on the type of pump used, primarily by the illustrations. Only very limited TM's actually specified pump manufacturer and model number information. Also, limited TM's provided hydraulic schematics.

A local Army Reserve depot was visited to gather information on the available vehicles and their hydraulic systems.

3.2 HYDRAULIC SYSTEM SURVEY

Two hydraulic pump manufacturers were contacted that were known to be suppliers of pumps for these vehicles. Information was requested on the tolerance of the pumps to fluids at low temperatures and high viscosity. However, the only information that was received was references to standard catalogue guidelines and specifications. Table 2 is a summary of typical fluid viscosity recommendations by several pump manufacturers and for different types of pumps.

Table 2. Pump Viscosity Recommendations

Hydraulic Pump Fluid Viscosity Requirement (cSt)			
Pump	Minimum	Optimum	Maximum
Rexroth (A10 Piston Pump)	5	16-36	1600
Parker (G Gear Pump)	7.5-10	>20	1600
Parker (PGP300 Gear Pump)	7.5	15-75	1600
Sauer (D Gear Pump)	10	12-60	1600
Sauer (40, 42 Piston Pump)	7	12-60	1600
Eaton (420 Piston Pump)	6-10	16-40	2100
Eaton (PVH Piston Pump)*	**	16-40	1000
Eaton (PVM Piston Pump)*	10	16-40	5000
Eaton (70XXX and PVE Piston Pump)*	6	10-39	432

** No data available.

A typical maximum viscosity specification for is 1600 cSt for several of the pumps. The specified viscosity for the previous Arctic MIL-PRF-46167C lubricant is 15,000 cSt at -40 °F, which is almost 10 times higher than what is recommended.

The recommended maximum fluid viscosity for most hydraulic valves, including directional, servo, and cartridge valves is usually less than 500 cSt, lower than for hydraulic pumps, requiring warmer fluid for proper operation.

The review of Army vehicles showed that their types of systems were similar to commercial vehicles. These systems can be classified in two different ways, open-center and closed-center, which refers to the state of the control valve(s) when in the neutral position.

Open-center systems are generally simpler and less expensive, using a fixed displacement pump, such as a gear pump or vane pump. A fixed displacement pump outputs fluid at a flowrate proportional to the engine speed. When the control valve(s) are in neutral, the flow passes through the 'open-center' unrestricted and at low pressure back to the reservoir. When the valve is actuated to operate a hydraulic cylinder, the flow is diverted to the actuator and builds up pressure to move the load. If the pressure level reaches the maximum rated system pressure, excess flow will pass over a relief valve back to the reservoir.

Closed-center systems use directional control valves in which the flow through the control valves is blocked when they are in the neutral position. A variable displacement pump is typically used with a pressure compensator control which regulates the displacement of the pump so that the maximum rated pressure is maintained, unless the pump has reached its maximum displacement. When the control valve(s) is in the neutral position no flow is required, so the pump displacement is regulated to a small amount, just enough to maintain the rated pressure compensated pressure setting. When the directional control valve is opened, flow passes to the actuator and the pump displacement increases to satisfy the flow requirement at the pressure compensated pressure level. When the pump reaches its maximum flow rate, the pump pressure will drop down to whatever is required to move the load plus any valve restrictions. This type of system is generally more efficient than an open-center system because less energy is lost since there is rarely flow over a safety relief valve, but it is more expensive due to the more complex variable displacement pump.

A variation of the closed-center system is referred to either as a "load-sensing" system or a "pressure and flow compensated" system. This system uses a directional control valve that sends a pilot hydraulic signal back to the pump so that the pump can sense the actual pressure required to move the load. The pump uses a load sensing displacement control valve that adjusts the pump displacement to regulate the pressure to approximately 300 psi higher than the load pressure. When the directional control valve is in the neutral position, the pilot load sensing line is vented to the reservoir, so the pump pressures is controlled down to load sensing level of approximately 300 psi. This type of system is the most efficient because it reduces the pressure drop across the directional control valve when modulating the flow to move a load, thus reducing energy loss. It is also slightly more expensive than a pressure compensated system because of the increased complexity.

3.3 TARGET VEHICLE SELECTION

After completing the vehicle survey, the 6K Rough Terrain Forklift, NSN 3930-01-158-0849 was selected as the target system. The manufacturer designates this military vehicle as a Skytrak 6000M. Throughout this report, the vehicle will be referred to as the 6000M forklift except where referring to it as a 6K Rough Terrain Forklift would denote the specific vehicle in the military fleet. Although this is not a high density vehicle within the Army fleet, this vehicle has a hydraulic system with characteristics common to most vehicles studied. It uses a tandem gear pump for steering, braking, and forklift functions operating in an open-center system. It also uses a variable displacement piston pump primarily for boom end-effecter control, which uses a closed-center load-sensing system. In addition to incorporating common hydraulic components, this vehicle was available at a nearby Army Reserve unit for detailed inspection. Additionally, the pumps and inlet screens were commercially available through a forklift supplier. Figure 1, 6K Rough Terrain Forklift, depicts the vehicle chosen for this work.

Figure 1. 6k Rough Terrain Forklift

After choosing the vehicle, we returned to the Army Reserve unit to inspect the vehicle in more detail. We photographed the hydraulic system in detail to facilitate reconstruction of the reservoir, pump inlet lines and pumps in the test chamber. We measured inlet hose lengths and diameters and noted number and type of hose fittings. We measured pump height in relation to the full mark on the hydraulic oil reservoir. We took notes of pump orientation, part numbers, and control lines.

With photographs, measurements and part numbers in hand, we contacted forklift dealers to locate and order specific parts for the test. In particular, we needed inlet screens and hydraulic pumps that duplicated those on the 6000M. Additionally, we needed a hydraulic schematic of the overall system to assure we were duplicating the vehicle system characteristics. We located a forklift dealer who could supply the parts and had a relationship with the manufacturer (Skytrak) to supply the hydraulic schematic. We ordered the inlet screens and pumps and began fabrication of the test stand to mount them. Figure 2, 6000M Forklift Hydraulic Schematic, depicts the manufactures hydraulic schematic for the forklift.

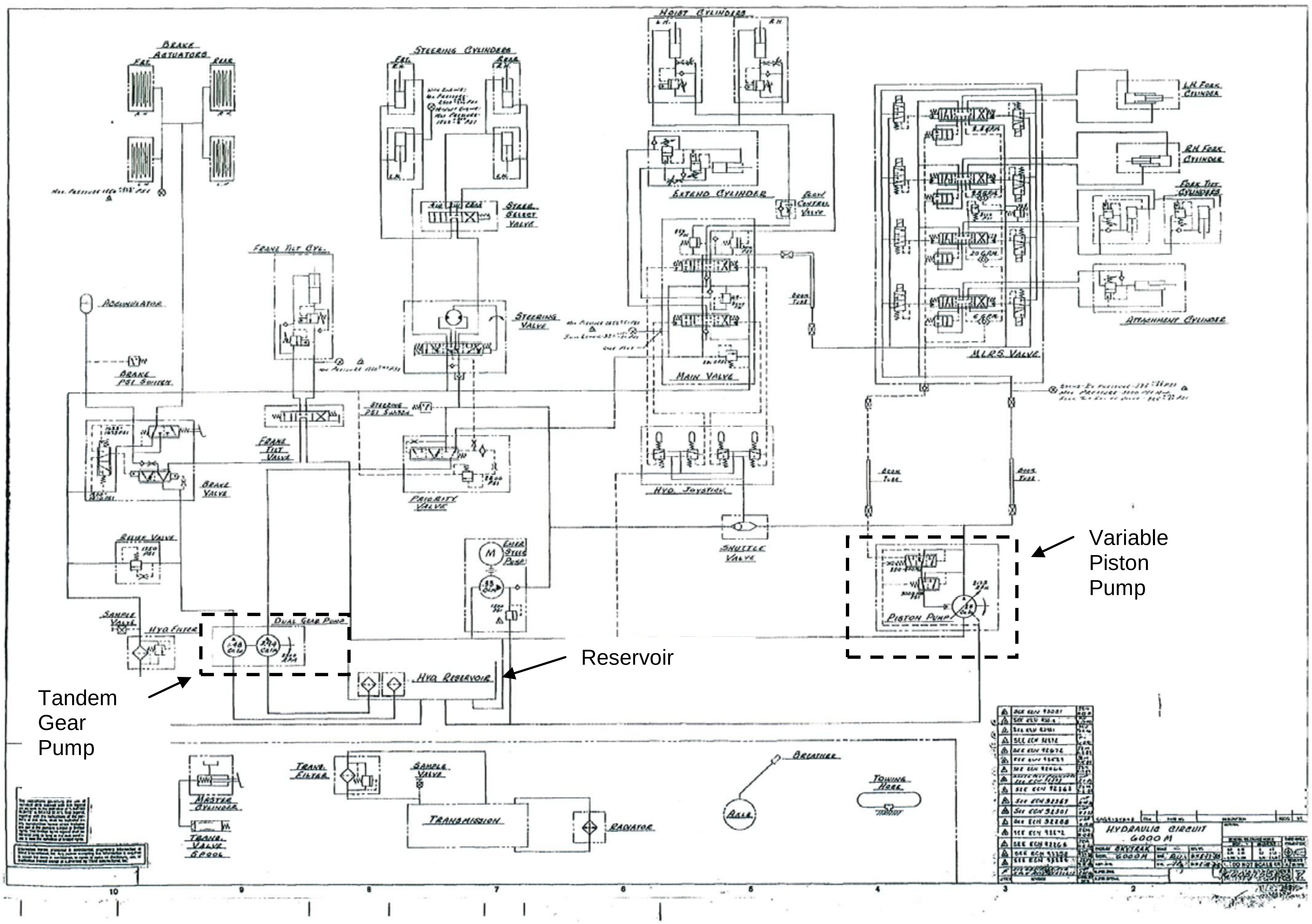

Figure 2. 6000M Forklift Hydraulic Schematic

4.0 TEST SYSTEM SETUP

The specification for the test stand was that it be able to rotate the test articles at speeds representative of vehicle idle, which is the most critical condition of a cold startup. Until a vehicle hydraulic system can operate at idle, increasing engine speed will only increase the level of cavitation and increase the risk of component damage. Skytrak reported that the idle speed for the forklift was 950 rpm and that the hydraulic pumps rotated at engine speed. We designated a 20 horsepower variable speed motor and drive unit to power the test apparatus. Calculations of the hydraulic pumps at the designated speed, pressures and flows indicated that the drive system would have adequate torque to rotate the pumps. The test system components consisted of the variable speed drive motor, inline torque meter, couplings, pump support bracket, inlet screens, hydraulic pumps, ball valve, relief valve, positive displacement flow meter, hydraulic reservoir and hoses and fittings to connect the components. Table 3, Test Components, describes the major test components.

Table 3. Test Components

Item	Description
Drive Motor	Marathon Electric, 256T Frame, 20 HP, 3500 rpm
Variable Speed Drive	Magnatek, Model GPO 503
Couplings	Magnalloy, M500 Series
Torque Meter	Key Mod. 4105-01 (2000 in. lb, Accuracy: ± 1% FS.), S/N 163D, SwRI 03-700873
Foot Mount	Vescor, Model FM350, 2 & 4 Bolt SAE-B
Inlet Screen for Fixed Displacement	Skytrak 8324013
Inlet Screen for Variable Displacement	Skytrak 8324011
Fixed Displacement Pump Assembly	Parker, Part Number 324-9120-075, SN N0106-08602
Variable Displacement Pump	Vickers/Eaton PVE19AR05AA10B2124000100100CDOAC
Ball Valve	Hycon Model KHB, 1 ¼”
Relief Valve	Sun RDHA-LCN
Positive Displacement Flow Meter	Kuppers, Model ZHM05, SN 02729508, SwRI 03-700975
Hydraulic Reservoir	Vescor Model 215149, Vertical, 30 Gallon, Removable Top
Pressure Transducers	Sensotec, 500 psi ± 5 psi
Temperature	Thermocouple, J-Type, ± 1 °F

4.1 FIXED DISPLACEMENT PUMP TEST SETUP

Inlet screens, hoses, and fittings on the inlet side of the pumps were carefully selected to exactly replicate the lengths, diameters and geometry on the 6K rough terrain forklift. A 1.5 inch diameter suction hose was used for the large, fixed displacement pump and a 1.0 inch diameter hose was used for the smaller fixed displacement pump. Fluid level in the test system reservoir was adjusted to match the vertical relationship between the forklift reservoir full mark and the pump inlet fittings. The complete test stand was located inside a cold box to simulate cold weather startup environment. The discharge circuit of the system was configured to be a simplified replication of the vehicle circuit. In the case of the tandem gear pump, the two discharge lines were joined with a "T" and the combined flow was measured by a single flow meter. Figure 3, Fixed Displacement Tandem Pump Hydraulic Schematic, illustrates the hydraulic schematic for the tandem pump test setup. The hydraulic schematic in Figure 2 indicates that the two pumps are 1.48 cu in./rev (24.2cc/rev). and 3.94 cu in./rev (64.5 cc/rev). for a total displacement of 5.42 cu in./rev (88.8 cc/rev). At the test speed of 950 rpm the theoretical flow rate of each pump is 6.1 gpm (23 lpm) and 16.2 gpm (61.3 lpm), respectively, for a total flow rate of 22.3 gpm (84.3 lpm). Twenty seven percent of the total theoretical flow capacity is with the small pump.

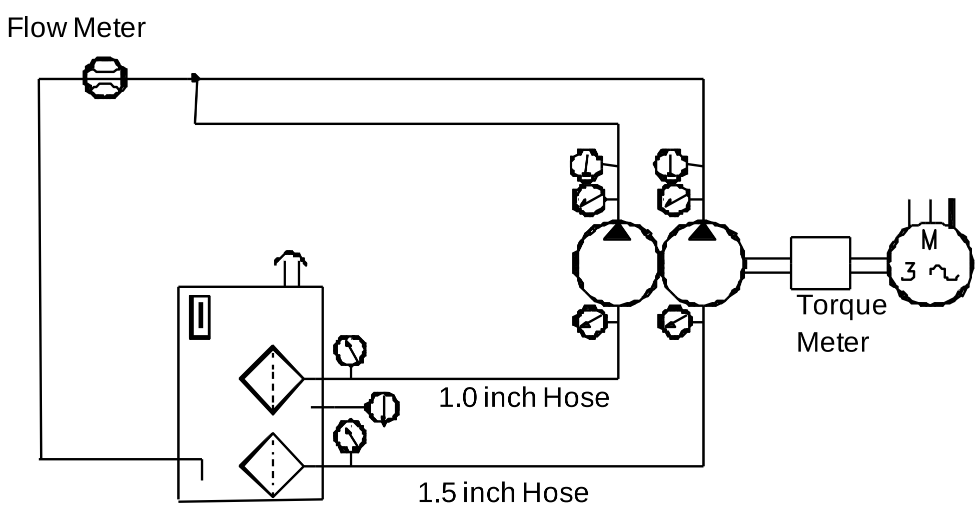

Figure 3. Fixed Displacement Tandem Pump Hydraulic Schematic

Figure 4, Fixed Displacement Tandem Pump Installation, depicts the test setup for the fixed displacement pumps installed on the test stand before pressure transducers were installed on the inlet and outlet fittings.

Figure 4. Fixed Displacement Tandem Pump Installation

4.2 VARIABLE DISPLACEMENT PUMP TEST SETUP

Inlet screens, hoses and fittings on the inlet side of the pumps were carefully selected to exactly replicate the lengths, diameters, and geometry on the 6K rough terrain forklift. Initially a ¾ inch diameter inlet hose was used to duplicate the inlet hose on the forklift. Initial testing indicated that this size inlet hose produced excessive pressure drop, even at moderate temperatures. The excessive pressure drop would effectively remove any discriminating ability from our tests. After further investigation it was determined that on the vehicle the pump inlet hose was connected with a tee fitting into the return line of one of the fixed displacement pump circuits and relying on the backpressure of the return line to provide additional flow to the variable displacement pump inlet. This configuration is not typical and not normally recommended. Based on this result, flow calculations and manufacturer recommended inlet hose size, we changed the inlet hose to 1.5 inch diameter. For the variable displacement pump, outlet flow was routed to a ball valve that was actuated externally to the cold box. The ball valve allowed us to simulate the vehicle starting with no load on the hydraulic system (closed center system, valve closed) and transition to a driver's command for hydraulic actuation by opening the ball valve after 30 seconds of operation. A pressure relief valve was added in parallel to the ball valve circuit to protect the system should the pump's internal flow and pressure regulation fail. The setting on the relief valve was set to be 3000 psig. The maximum displacement of the pump is 2.50 cu in./rev (41 cc/rev) and at the test speed of 950 rpm the theoretical flow is 10.3 gpm (38.9 lpm). Figure 5, Variable Displacement Pump Hydraulic Schematic, illustrates the hydraulic schematic for the variable displacement pump test setup. The load sensing pilot line was routed from the pump to the reservoir so that the regulated pressure would be at the load sensing pressure.

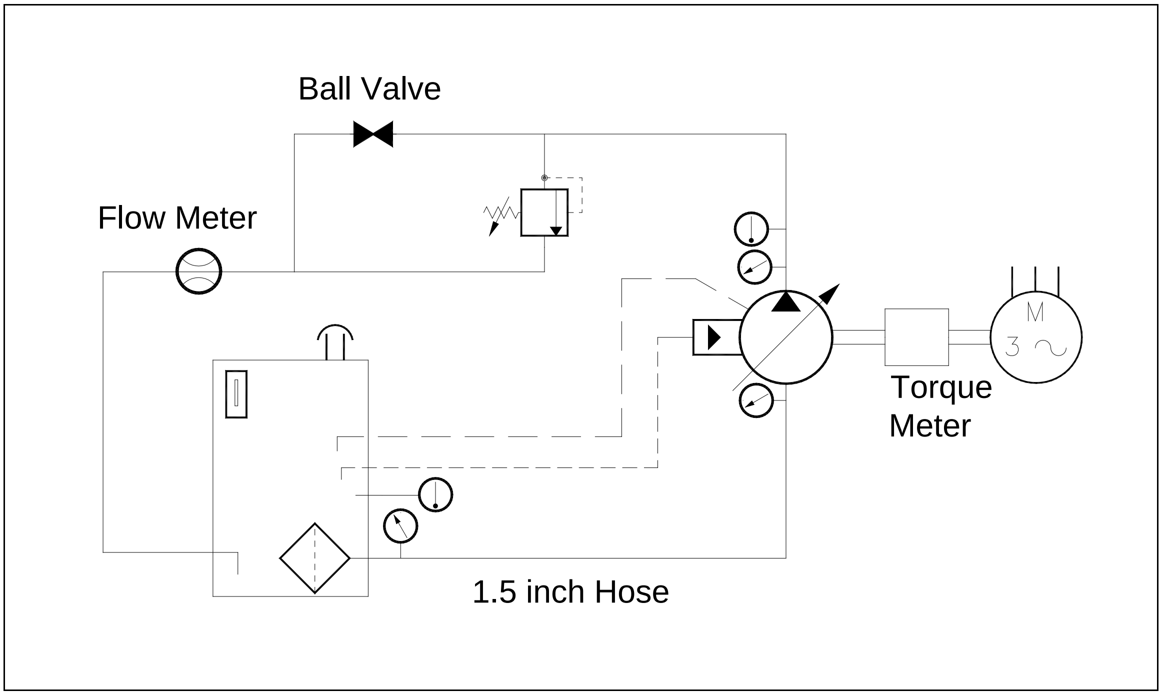

Figure 5. Variable Displacement Pump Hydraulic Schematic

Figure 6, Variable Displacement Pump Installation, depicts the test setup for the variable displacement pump installed on the test stand under cold and icy conditions.

Figure 6. Variable Displacement Pump Installation

4.3 TEST OILS

Three oils were used in these evaluations. The oils were AL-27637 (MIL-PRF-46167C), LO-228213 (MIL-PRF-46167D) and Mobil Univis HV-26. The Mobil Univis HV-26 is a premium performance anti-wear commercially available hydraulic oil that has an unusually high viscosity index (very flat viscosity versus temperature curve). Table 4, Kinematic Viscosity of Test Oils lists the kinematic viscosity of the test oils by ASTM D445 over a wide temperature range. Figure 7, Kinematic Viscosity of Test Oils graphically depicts the kinematic viscosity data for the test oils.

Table 4. Kinematic Viscosity of Test Oils

		UNIVIS HVI-26	LO-228213	AL-27637
			MIL-PRF-46167D	MIL-PRF-46167C
Temperature	Temp. Numeric	K. Vis. cSt	K. Vis. cSt	K. Vis. cSt
-40 °F	-40	1096.19	17901.41	15140.51
-20 °F	-20	442.61	5015.79	4182.77
0 °F	0	218.34	1767.48	1471.22
20 °F	20	124.51	734.02	609.29
40 °F	40	78.87	343.89	289.5
60 °F	60	53.46	178.9	152.41
100 °F	100	28.78	63.39	55.79
212 °F	212	9.47	10.59	9.98
140 °C (284 °F)	284	6.46	5.38	5.26

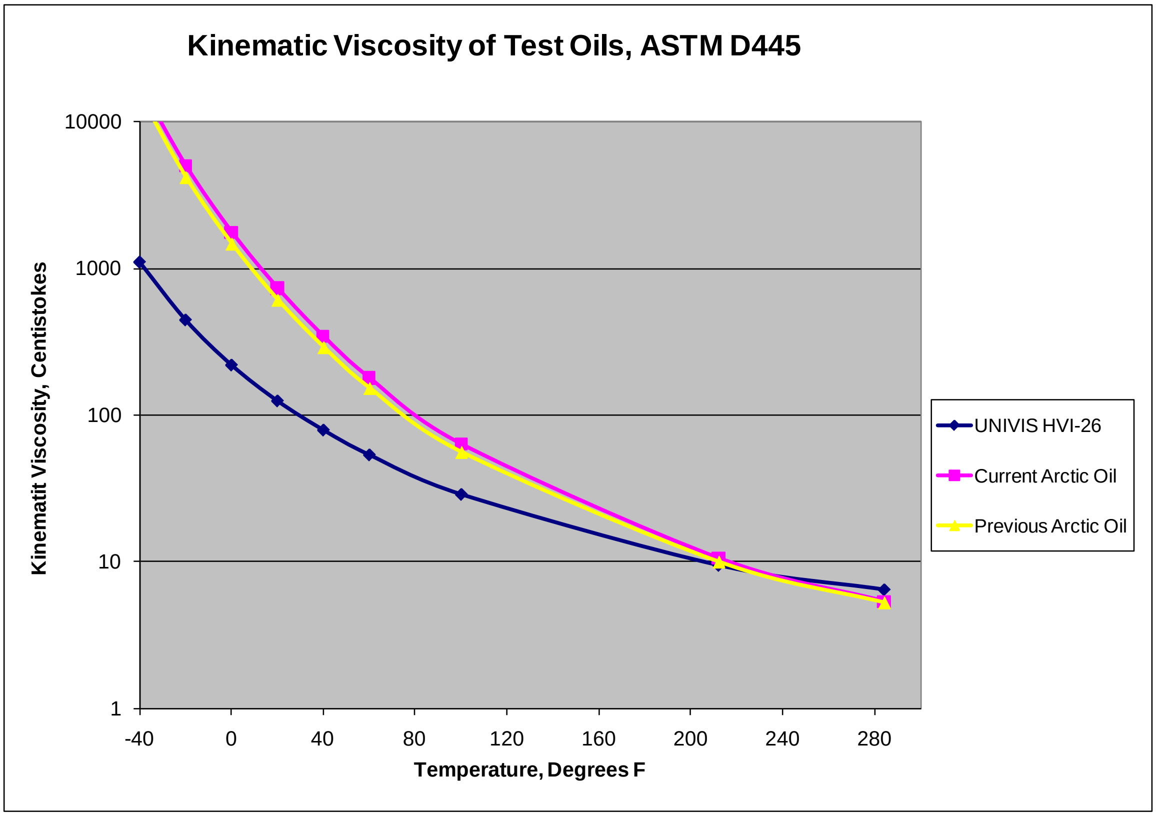

Figure 7. Kinematic Viscosity of Test Oils

The Mobil Univis Oil displays significantly lower viscosity at the lower temperatures and higher viscosity at temperatures above 210 ºF than the arctic oils. This indicates that the Univis Oil will maintain better volumetric efficiency at low temperatures than the arctic oils.

The recommended minimum operating temperature can be predicted for each pump based upon the minimum recommended inlet viscosity. Table 5 lists the recommended minimum viscosity for each pump and the temperature required for each fluid to have the respective viscosity, interpolated from the data obtained from testing the oils.

Table 5. Temperature of Fluid for Recommended Maximum Viscosity for Tested Pumps

Pump Type	Maximum Recommended Viscosity	Temperature, ºF		
		UNIVIS HVI 26	LO-228213 Current Arctic Oil	AL-27637 Previous Arctic Oil
Parker Tandem Gear Pump	1600 cSt	-48	2.1	-1.8
Eaton/Vickers PVE Axial Piston Pump	432 cSt	-19.3	34	33

4.4 TEST PROCEDURE

Cold pumping tests consisted of charging the system with the test oil, soaking the entire system to the desired test temperature for a period of approximately 23 hours, turning on a variable speed drive to a test article rotational speed of 950 rpm and recording data on system performance. Each test was run for 10 minutes, during which time the circulating oil slowly increased in temperature and flow rate. This procedure was repeated for different test temperatures and then a new test fluid was introduced into the system. Test temperatures were chosen based on how each fluid performed. Test temperatures represent initial reservoir temperature near the inlet screen. Two cold box defrost cycles were implemented during each 23 hour cold soak procedure, one at midnight and one at 5 am to minimize frost on the condenser coils and assure desired test temperatures were reached. Appendix A, Lubricant Pumpability Test Procedure, describes the test procedure used for fixed and variable displacement tests.

Appendix B, Oil Drain, Flush and Fill Procedures for Cold Lubricant Pumpability Tests, describes the oil drain, flush and fill procedures used for all test series. For each test the pump speed was ramped up to a steady 950 rpm within about two seconds.

4.5 TEST MATRIX

Test temperature represents the initial temperature of the oil in the reservoir, as measured close to the inlet screens just before each test run. One run with each geometry and oil was done at room temperature to verify room temperature oil performance and verify test system performance. Test temperatures were lowered at approximately 10 ºF increments. Minimum test temperatures were limited by cold box thermodynamics given ambient temperature and atmospheric moisture levels.

The fixed displacement tandem gear pump was run with oil AL-27637 (MIL-PRF-46167C Arctic Oil) at reservoir initial test temperatures of 41, 31, 22, 11, 1, -9, -22 and -30 ºF; with oil LO-228213(MIL-PRF-46167D Arctic Oil) at test temperatures of 68, 41, 30, 20, 11, -1, -11, and -19 ºF; and with oil Univis HV-26 at test temperatures of -9, -20, -30 and -32 ºF.

The variable displacement pump was run with oil AL-27637 (MIL-PRF-46167C Arctic Oil) at reservoir initial test temperatures of 74, 30, 21, 10, 0, -11, -19, and -25 ºF; with oil LO-228213(MIL-PRF-46167D Arctic Oil) at test temperatures of 75, 31, 20, 10, 1, -10, -21, and -23 ºF; and with oil Univis HV-26 at test temperatures of 86, 31, 20, 9, 1, -10, -18, -20, and -24 ºF.

5.0 DISCUSSION OF RESULTS FOR FIXED DISPLACEMENT TANDEM PUMPS

Figure 8, Typical Tandem Gear Pump Raw Data, depicts all measurements that were recorded for fixed displacement pump tests versus time for test oil AL-27637.

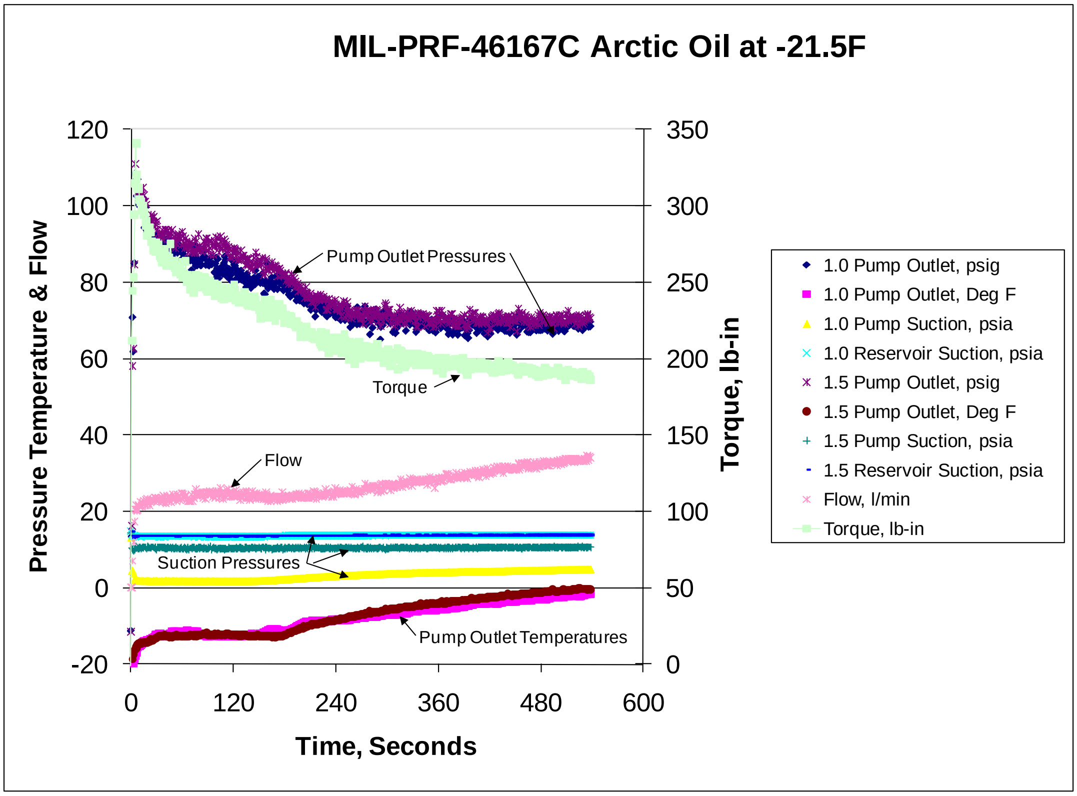

Figure 8. Typical Tandem Gear Pump Raw Data

Note that pump outlet temperature remains stable for approximately 180 seconds into the run and then begins to rise slowly. Outlet flow follows the temperature trend, rising slightly at first, then remaining relatively stable, then rising slowly after about 180 seconds. Torque starts off high and drops rapidly as friction warms the pumps. Outlet pump pressures for both the 1 inch and 1.5 inch diameter lines start out high and drop slowly as the circulating oil warms. Pump suction pressures and reservoir suction pressures (in psia) stay relatively constant throughout the test. From a system performance standpoint, the ability of the hydraulic system to deliver flow is a good metric of its efficacy. Flow in this test apparatus is largely dominated by the pump suction pressures, which in turn is affected by inlet screen restriction, fitting geometry, hose diameter, oil viscosity. and hose length. For this reason, we will concentrate on suction pressures and flow data throughout the majority of this discussion.

Figure 9, Summary Data for MIL-PRF-46167C Arctic Oil in Tandem Gear Pumps, depicts suction pressures and combined flow of one oil, under the entire slate of test temperatures.

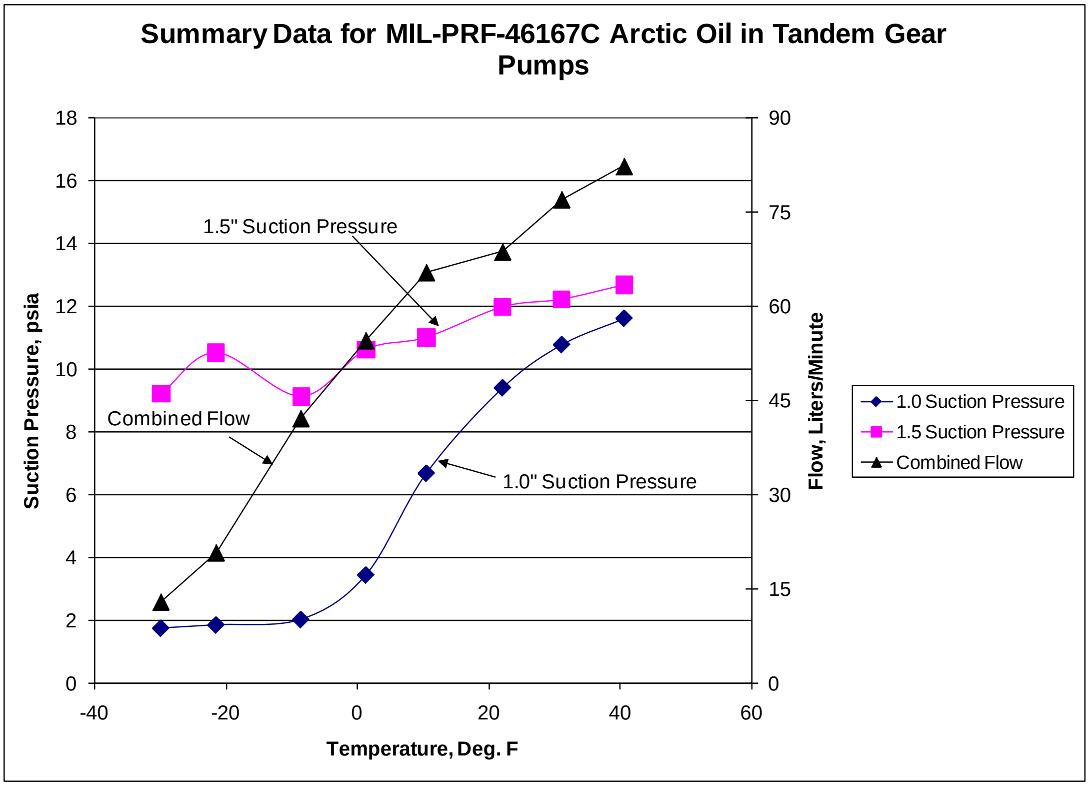

Figure 9. Summary Data for MIL-PRF-46167C Arctic Oil in Tandem Gear Pumps

Data plotted in Figure 9 represent the average of 5 readings taken at the beginning of each test run after stable pump speed was achieved. As test temperature decreases, suction pressure for the 1 inch pump drops off more steeply than for the 1.5 inch pump, primarily due to the reduced size of the suction hose. Even near 40 °F, the suction pressures of the pumps had begun to decline precipitously from atmospheric pressure (14.7 psia) with a corresponding drop in combined flow. The low suction pressures have starved the pumps of inlet flow, reducing their volumetric efficiency.

Figure 10, Summary Data for MIL-PRF-46167D Arctic Oil in Tandem Gear Pumps, depicts the suction pressures and combined flow under the entire slate of test temperatures.

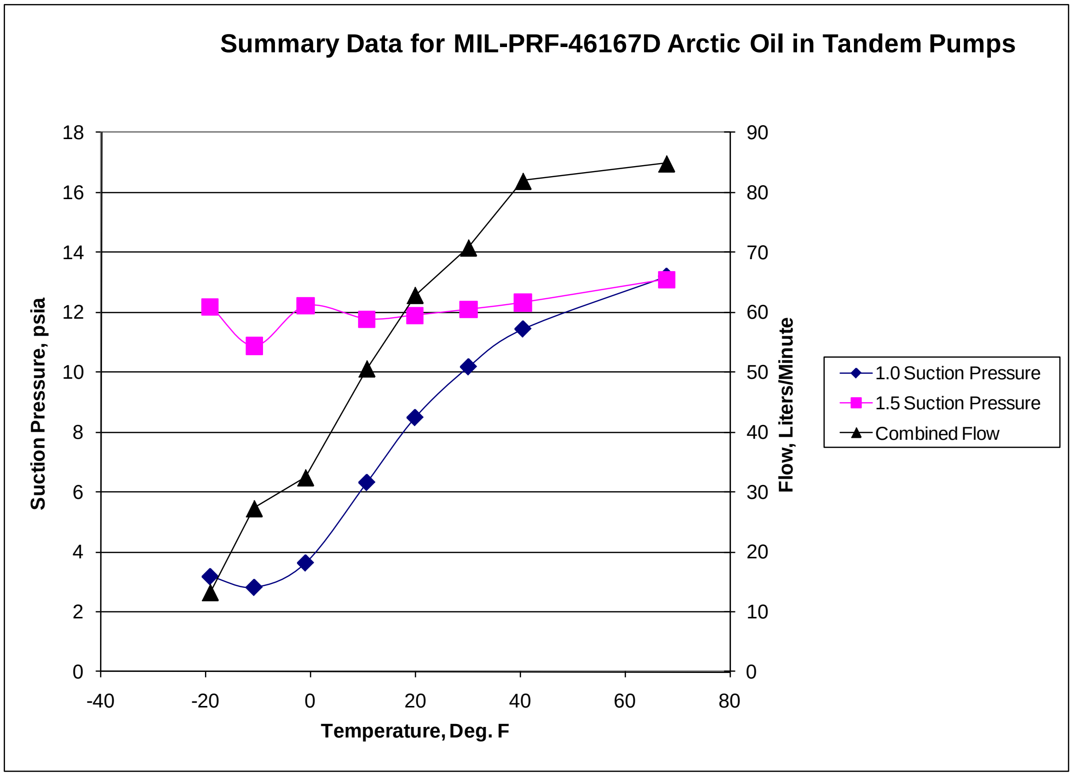

Figure 10. Summary Data for Current Arctic Oil in Tandem Gear Pumps

In Figure 10, once again, as test temperature decreases, suction pressure for the 1 inch pump drops off more steeply than for the 1.5 inch. Beginning at 40 ºF, the suction pressures of the pumps has begun to decline from atmospheric pressure with a corresponding drop in combined flow. As in Figure 9, low suction pressures have starved the pumps of inlet flow, reducing their volumetric efficiency.

Figure 11, Summary Data for Univis in Tandem Gear Pumps, depicts the suction pressures and combined flow of the Mobil Univis oil under the entire slate of test temperatures.

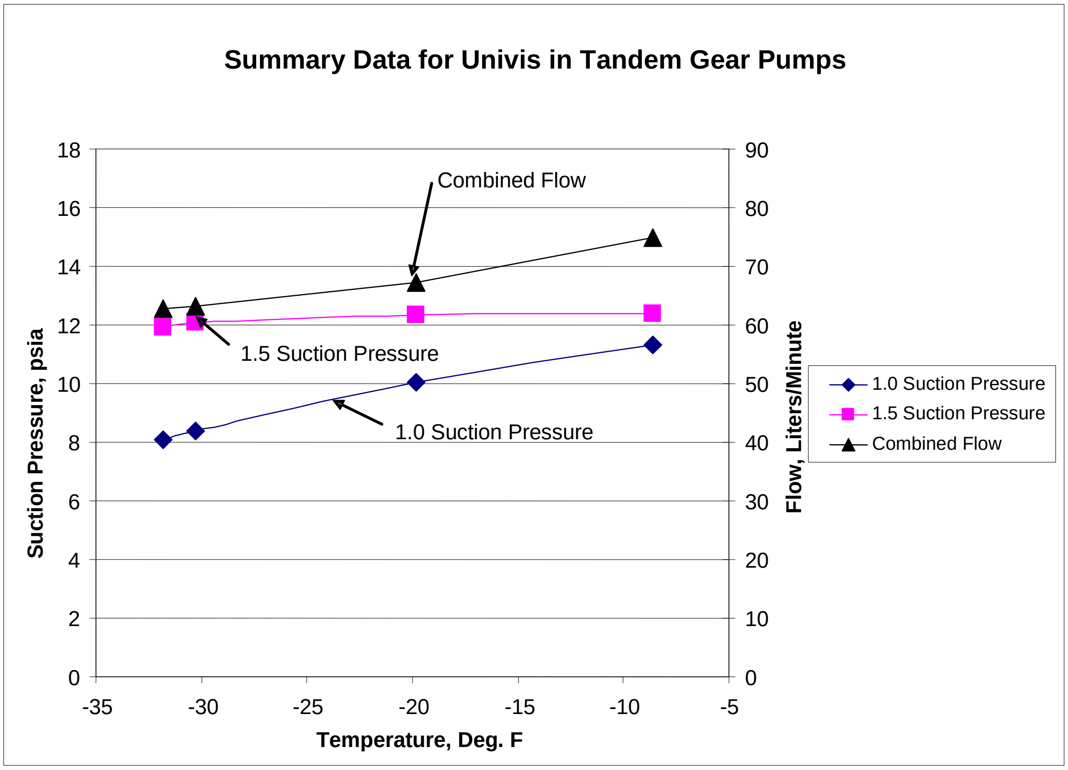

Figure 11. Summary Data for Univis in Tandem Gear Pumps

In Figure 11, suction pressures drop off less rapidly as a function of temperature than the previous Arctic Oil depicted in Figure 9. As a result, combined flow decreases less rapidly. This is primarily a function of the lower viscosity of the Univis oil in comparison to the other test oils. Refer to Figure 7, Kinematic Viscosity of Test Oils to compare kinematic viscosity of the test oils.

Figure 12, Summary Combined Flow Data for All Oils – Fixed Displacement Pumps, depicts flow data for all test oils under all temperature conditions. Again, flow data represents the average of five readings taken at the beginning of each test run after stable pump speed was been achieved.

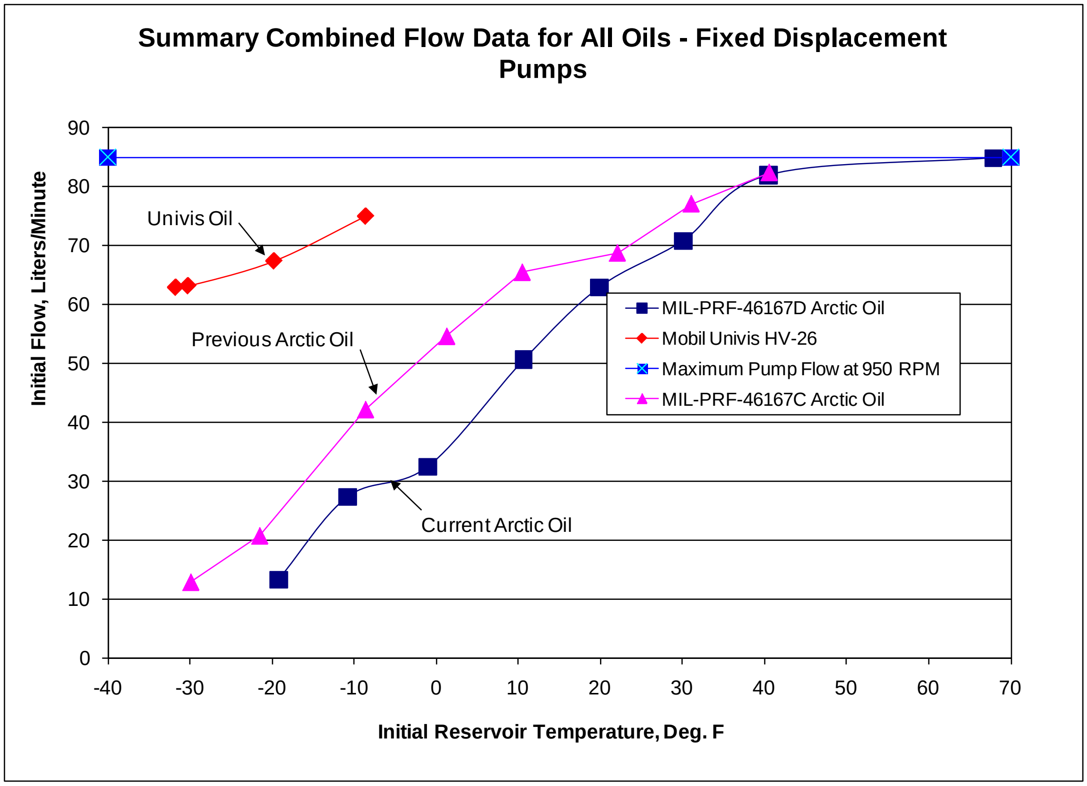

Figure 12. Summary Combined Flow Data for All Oils – Fixed Displacement Pumps

In Figure 12, note that the previous arctic oil maintains a higher flow at a given temperature than the current arctic oil. Also note that the Univis oil maintains a much higher flow than either of the two arctic oils. Maximum calculated pump flow, based on the rated volume per revolution and speed is shown in Figure 12 as a line at 85 liters per minute. Lower hydraulic flow relates to a vehicles ability to respond rapidly to an operators request for a particular function and not feel “sluggish”. Since the fixed displacement pumps power the braking and steering on the 6000M forklift, operators could feel sluggish steering or braking under cold conditions if the vehicle were required to start and operate quickly. Many operators of vehicles in cold climates, however, start vehicles well before they need to begin operations. Many operators leave vehicles idling during extreme cold weather to warm the hydraulic oil by circulation and ameliorate such cold-related problems. Operators using the Univis oil would experience fewer instances of sluggish performance than those using the previous arctic oil or current arctic oil For a given required flow, the current arctic oil appears to be 5 to 10 °F more severe than the previous arctic oil.

These comparative results are very good at discriminating the pumping characteristics of oils relative to other oils. However, it is difficult to extrapolate this data to actual performance of a hydraulic oil in a vehicle, and much more difficult to extrapolate performance of a given oil in the military fleet. This is due to the wide range of operational differences between vehicles within the military fleet and the wide range of pump inlet geometries present within different classes of vehicles within the fleet.

Upon reviewing the test results, it is apparent that the inlet flow restriction of the smaller pump is much greater than for the larger pump because the suction pressure on the 1 inch line drops very quickly at colder temperatures. The equations below will help understand the sensitivity of hose diameter on fluid flow capacity.

$$Q = \frac{\pi D^4}{128 \mu L} \Delta P$$

Where:

Q = *Flowrate*

D = *Inside Diameter of line*

μ = *Fluid Absolute Viscosity*

L = *Length of line*

ΔP = *Pressure drop across line*

The flow rate through a line, assuming laminar flow, is directly proportional to pressure drop and inversely proportional to fluid viscosity and the length of the line. With all other variables held constant, the flow rate will vary with the inside diameter of the line raised to the fourth power. If we compare a 1.5 inch hose (which is the nominal inside diameter) to a 1 inch hose, the 1.5 inch hose will have a 5.1 times higher flow capacity. The larger pump is connected to the 1.5 inch hose and it is 2.6 times larger. The larger pump has an inlet line that has approximately twice the flow capacity as compared to the pump size, so one would expect it to have less sensitivity at cold temperatures.

Figures 10 and 11 reveal that the suction pressure of the 1.5 inch hose drops very little with colder fluid temperature, so one would expect that there should be little starving of the large pump. The suction pressure of the 1.0 inch hose, on the other hand drops dramatically with colder fluid temperatures. The flow rate, however, also drops off dramatically at cold temperatures. In Figure 11, for the current arctic fluid, the combined flow at -20 ºF is approximately 12 lpm, or approximately 86 percent less than the theoretical flow. This reduction in flow indicates that there was starving of the larger pump at colder temperatures, even though the suction pressure remained relatively high. This suggests that the restriction in the suction line was not the dominant restriction, but that the internal restriction from the pump inlet port and the pumping cavity is the dominant restriction. The region of highest restriction is probably the entrance to the pumping cavity itself, which is the same cross-sectional flow area as the smaller pump. The difference in the geometry of the small pump versus the large pump is only in the width of the gear. As fluid enters the cavity opened by the gears, it has to travel from one side of the gears to the opposite side to fill the cavity within a short amount of time. On the large pump, this distance is about twice as far as on the small pump; consequently, as the fluid gets more viscous, less volume is able to flow into the pumping cavity.

6.0 DISCUSSION OF RESULTS FOR VARIABLE DISPLACEMENT PUMP

Figure 13, Typical Variable Pump Raw Data, depicts all variables that were recorded for variable displacement pump tests versus time for test oil AL-27637.

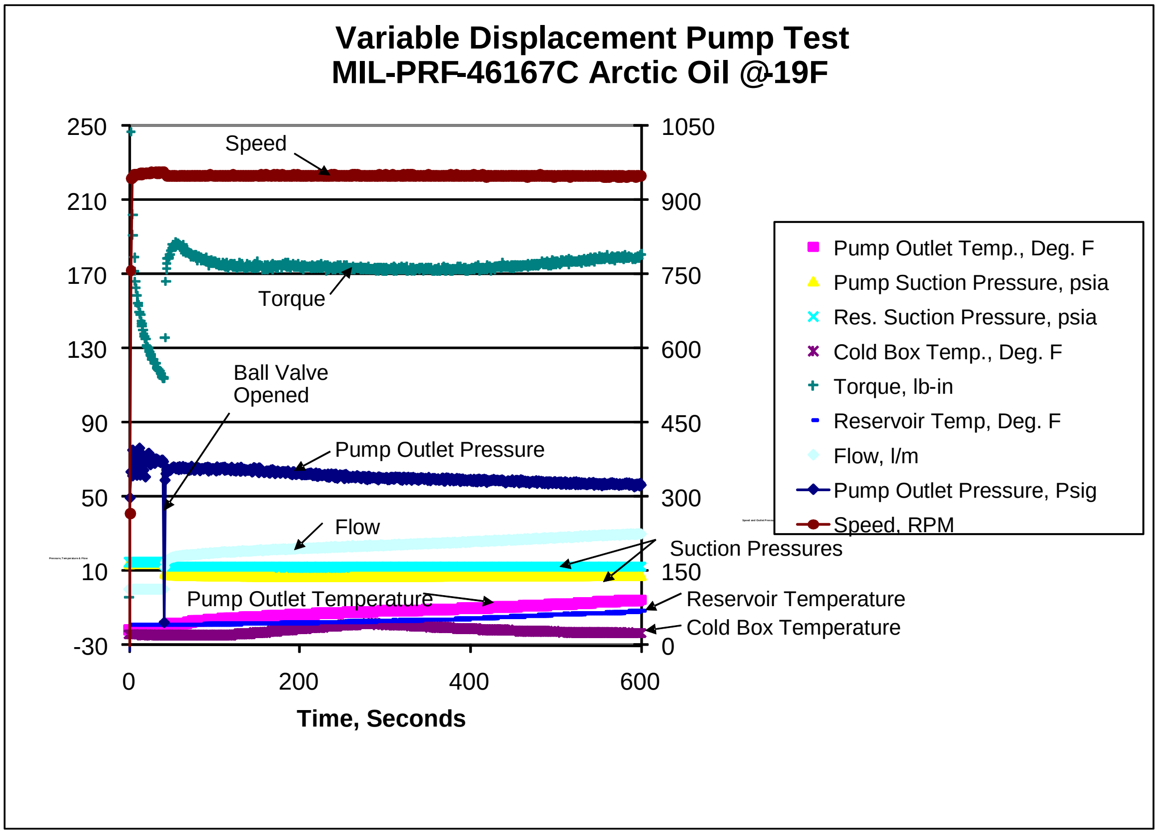

Figure 13. Typical Variable Pump Raw Data

In Figure 13, note that the test procedure shifts from zero flow to maximum flow at approximately 30 seconds of test time when the ball valve is opened. After the valve shift, flow quickly stabilizes and begins to increase slowly, paralleling pump outlet temperature. Again, oil flow in this test apparatus is largely dominated by the pump suction pressures, which in turn is affected by inlet screen restrictions, fittings geometry, hose diameter, oil viscosity, and hose length. Pump outlet temperature increases slowly as does reservoir temperature. Both pump and reservoir suction pressures remain relatively constant over time. Torque starts off high, drops off rapidly as the stagnated oil in the pump increases in temperature, then rises rapidly as the oil begins to flow. Pump outlet pressure drops off slowly as the circulating oil warms.

Figure 14, Summary Data for MIL-PRF-46167C Arctic Oil in Variable Displacement Pump, depicts pump suction pressure and pump flow of one oil, AL-27637, previous arctic oil, under the entire slate of test temperatures.

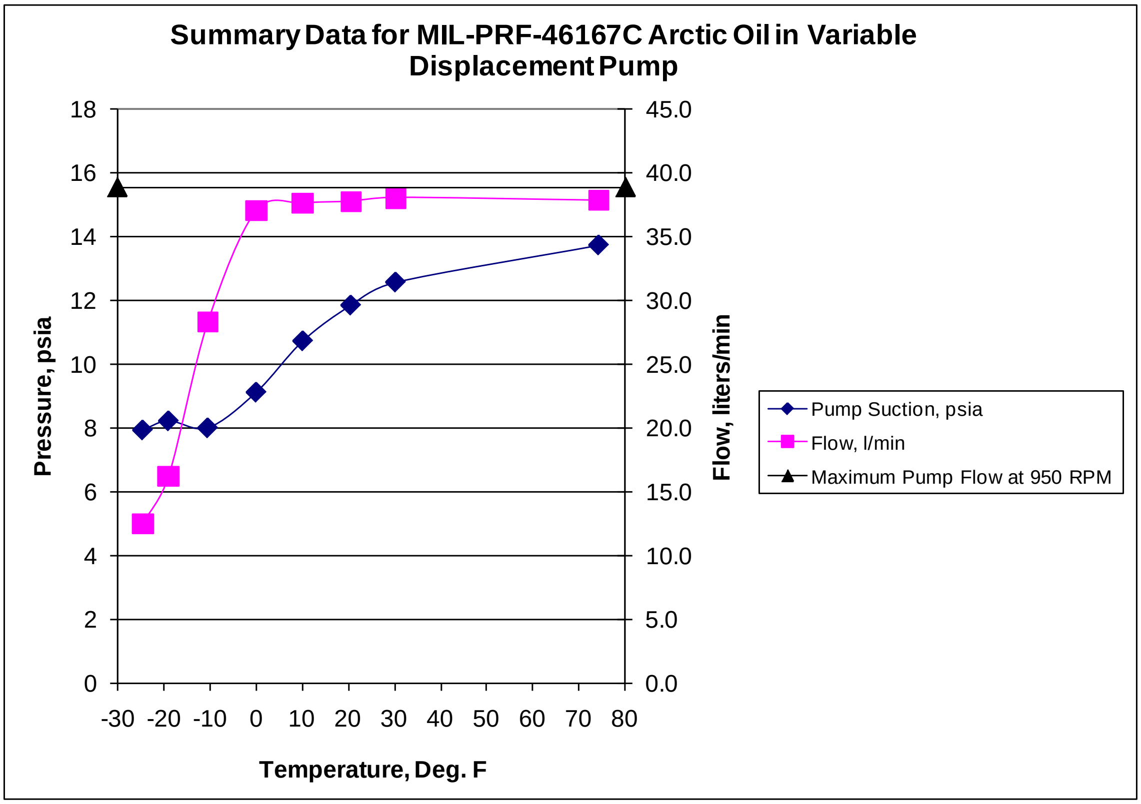

Figure 14. Summary Data for MIL-PRF-46167C Arctic Oil in Variable Displacement Pump

In Figure 14, a line representing maximum calculated pump flow at 950 rpm is set at 38.9 liters per minute. This is the theoretical flow the pump should achieve under ideal maximum displacement conditions at 950 rpm. Flow and suction pressure data represent the average of five readings taken at the beginning of each flow portion of each test, after stable pump speed was achieved and after the ball valve had been opened.

In Figure 14 notice that as temperature drops between 0 °F and -10 °F the flow begins to reduce significantly. Also notice that at -10 °F the suction pressure stops dropping and remains approximately 8 psia, almost one half of atmospheric pressure, for the next two lower temperature runs while flow continues to drop. This implies that the restriction between the reservoir and the pressure transducer on the fitting at the pump suction port, is equivalent to the restriction between the pressure transducer and the pumping cavity.

Figure 15, Summary Data for current arctic oil in Variable Displacement Pump, depicts pump suction pressure and pump flow of one oil, LO-228213, current arctic oil, under the entire slate of test temperatures.

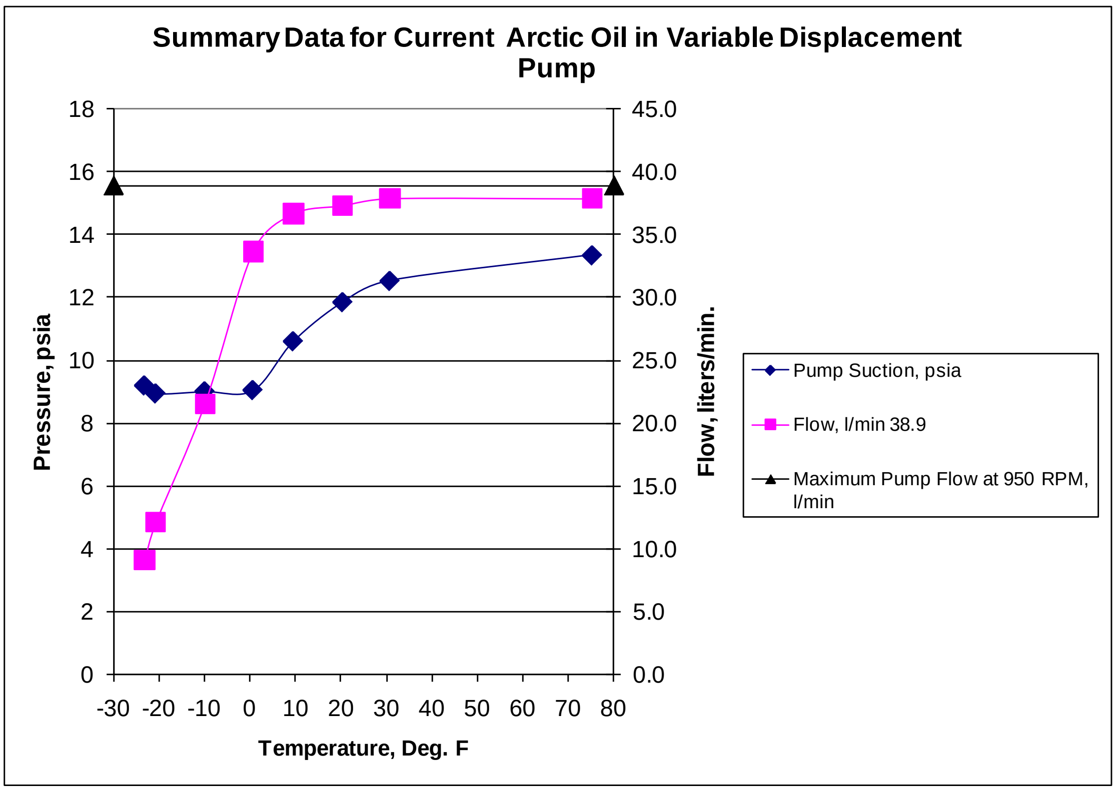

Figure 15. Summary Data for Current Arctic Oil in Variable Displacement Pump

In Figure 15, with the current arctic oil, notice that a similar characteristic is shown as in Figure 14 with the previous arctic oil, except at approximately 5 °F warmer temperatures. The flow begins to drop as temperatures are lowered below 0 °F, and the suction pressure remaining constant, at approximately 9 psia.

Figure 16, Summary Data for Univis Oil in Variable Displacement Pump, depicts pump suction pressure and pump flow of the Mobil Univis HV-26 Oil, under the entire slate of test temperatures.

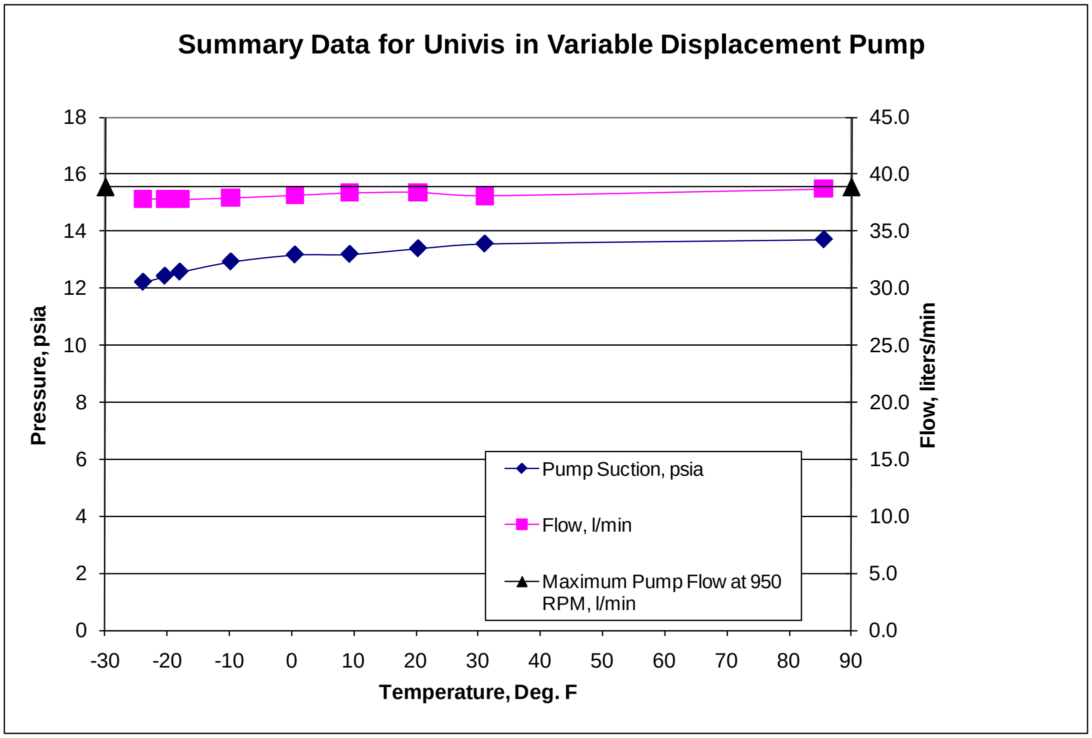

Figure 16. Summary Data for Univis Oil in Variable Displacement Pump

In Figure 16, it is noted that the pump suction pressure does not drop as steeply with decreases in temperature as the Arctic oils. The temperature was never cold enough to result in a suction pressure as low as 9 psia, where flow began to drop significantly for the other more viscous oils. This is primarily a viscosity effect, a consequence of the excellent low temperature viscometrics of the Univis oil. Because suction pressure does not drop off appreciably and the pump maintains good volumetric efficiency, the Univis oil comes close to delivering theoretical maximum pump flow throughout the test temperature range.

Figure 17, Summary Flow Data for All Oils – Variable Displacement Pump, depicts flow data for all test oils under all temperature conditions.

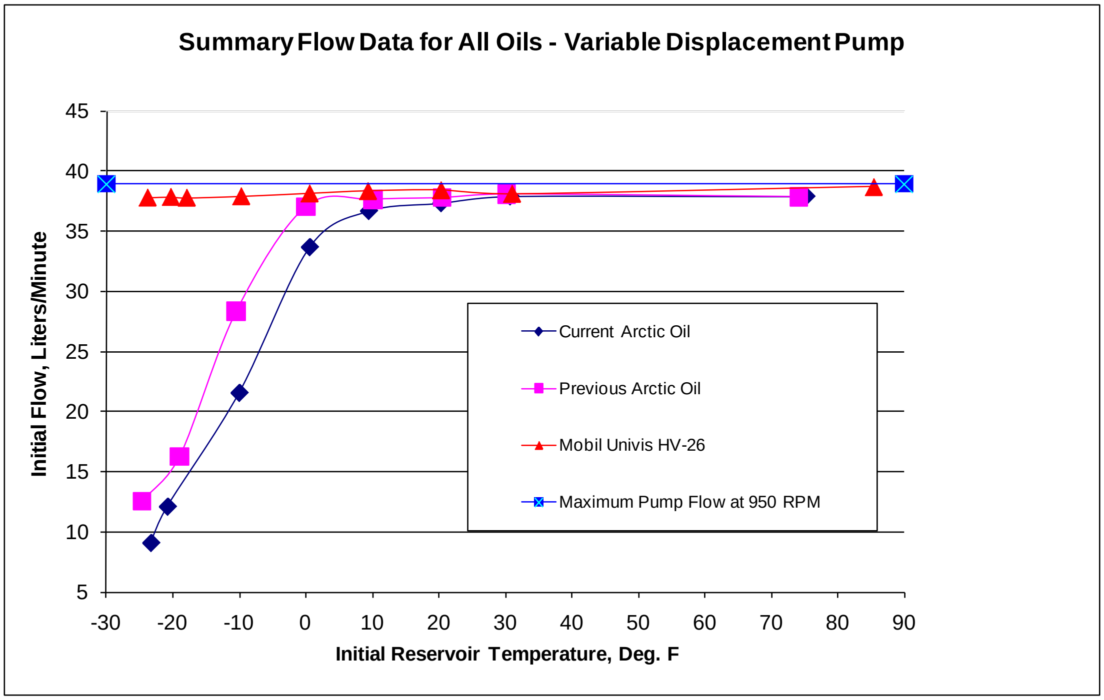

Figure 17. Summary Flow Data for All Oils - Variable Displacement Pump

As in the fixed displacement pump data, lower hydraulic flow relates to a vehicles ability to respond rapidly to an operators request for a particular function. Since the variable displacement pump powers the boom and end effecter on the 6000M fork lift, operators could feel sluggish load handling performance under cold conditions if the vehicle were required to start and operate quickly in cold climates. Many operators of vehicles in cold climates, however, start vehicles well before they need to begin operations. Many operators leave vehicles idling during extreme cold weather to warm the hydraulic oil by circulation. Operators using the Univis oil would experience fewer instances of sluggish performance than those using the previous arctic oil or current arctic oil. At ambient temperatures less than about 10 °F, for a given required flow, the current arctic oil appears to be 5 to 8 °F more severe than the previous arctic oil. The Univis oil would maintain performance at temperatures less than -24 °F.

7.0 CONCLUSIONS

The survey of military vehicles revealed a considerable number that use engine oil in the hydraulic system. Some vehicles used open center hydraulic systems with fixed displacement gear or vane pumps. Other vehicles used closed center hydraulic systems with variable displacement piston pumps.

A 6K Rough Terrain Forklift, NSN 3930-01-158-0849, was selected as the system target vehicle for duplication of the hydraulic system in an environmental chamber to simulate cold startup events and evaluate the pumpability of different oils.

The pumpability tests reported herein discriminate hydraulic oil pumpability characteristics well and allow comparisons to be made between candidate oils for the specific systems being tested.

As expected, the dedicated hydraulic fluid (i.e., Mobil Univis HV-26) performs well in both fixed displacement and variable displacement hydraulic pump systems. The Univis oil displays significantly lower viscosity at the lower temperatures than the arctic oils. This indicates that the Univis oil will maintain better volumetric efficiency at low temperatures than the arctic oils.

The current arctic oil has a slightly higher kinematic viscosity than the previous arctic oil and performed marginally poorer in the pumpability evaluations. The difference in pumping characteristics between current and previous arctic oils is approximately 5 to 10 ºF depending on the pump and its inlet characteristics.

It is difficult to extrapolate this data to actual performance of a hydraulic oil in a vehicle, and much more difficult to extrapolate performance of a given oil in the military fleet. This is due to the wide range of operational differences and the wide range of pump inlet geometries inlet screen restriction, fitting geometry, hose diameter, and hose length present within different classes of vehicles within the fleet.

The data presented herein indicates that when using multipurpose engine oils such as the arctic oil, hydraulic equipment, particularly those that are not designed to handle the higher viscosities, will respond sluggishly immediately after start-up. The severity of the response will be in proportion to the ambient temperature. Data collected from hydraulic pump manufacturers suggest that when oil viscosity is greater than 1600 cSt, pump performance will be significantly affected. Indeed, this behavior was confirmed in our laboratory simulated hydraulic system. That being said, the U.S. Army Tank Automotive Research Development & Engineering Center has no reported field problems with regard to the use of arctic oils in hydraulic systems. We must assume that our tests are either more severe than actual field service or operators have developed "work around" procedures such as starting equipment well in advance of actual need or continuously idling equipment in cold weather.

High viscosity of hydraulic oils at low temperatures can have multiple effects in addition to the primary effect of low flow and sluggish operation of the hydraulic system. Other damaging effects and failures can occur to the pump, valves, and actuators if these components are exposed to high speeds and high pressures. It is generally recommended that to avoid damage or failures in cold start-up conditions that the oil and the system be given time to warm up gradually.

APPENDIX A
LUBRICANT PUMPABILITY TEST PROCEDURE

Lubricant Pumpability Test Procedure

Tandem Pump Test Procedure

- Tandem gear pumps were installed on the test stand.
- Test oil was introduced into the system and the reservoir filled according to the “Oil Drain, Flush and Fill Procedures for Cold Lubricant Pumpability Tests” document.
- Test temperature was adjusted in the cold box. 23 hour soak times were used to assure equilibration.
- Data acquisition system was turned on to sample at 1 second intervals.
- Drive was preset to 950 rpm to simulate vehicle idle conditions.
- Drive was turned on and small adjustments made to reach 950 rpm.
- Parameters recorded were:
    1. Time, seconds
    2. Suction pressures near the inlet screens (1.0 and 1.5 pumps), psia
    3. Suction pressure at the pumps (1.0 and 1.5 pumps), psia
    4. Outlet pressure at the pumps (1.0 and 1.5 pumps), psig
    5. Output flow (both pumps combined), liters/minute
    6. Reservoir temperature, Degrees Fahrenheit
    7. Pump outlet temperature (1.0 and 1.5 pumps), Degrees Fahrenheit
    8. Cold box temperature, Degrees Fahrenheit
    9. Torque, lb-in
    10. Speed, rpm
- Each test lasted for 10 minutes, at which time the drive was turned off and data acquisition was terminated.
- Temperature of the cold box was re-adjusted for the next condition.
- Data was converted to Excel spreadsheet and plotted, emphasizing reduced flow rates as temperature drops.

At the completion of each lubricant temperature series, new oils were introduced into the system according to “Oil Drain, Flush and Fill Procedures for Cold Lubricant Pumpability Tests” document.

Variable Displacement Pump Procedure

- The variable displacement pump was installed on the test stand.
- Plumbing and a ball valve were added to the system.
- Test oil was introduced into the system and the reservoir filled according to the "Oil Drain, Flush and Fill Procedures for Cold Lubricant Pumpability Tests" document.
- Test temperature was adjusted in the cold box. 23 hour soak times were used to assure equilibration.
- Data acquisition system was turned on to sample at 1 second intervals.
- Drive was preset to 950 rpm to simulate vehicle idle conditions.
- The ball valve was initially positioned to block flow from the pump. This simulates closed center forklift operation.
- Drive was turned on and small adjustments made to reach 950 rpm.
- Parameters recorded were:
    1. Time, seconds
    2. Suction pressures near the inlet screens (1.0 and 1.5 pumps), psia
    3. Suction pressure at the pumps (1.0 and 1.5 pumps), psia
    4. Outlet pressure at the pumps (1.0 and 1.5 pumps), psig
    5. Output flow (both pumps combined), liters/minute
    6. Reservoir temperature, Degrees Fahrenheit
    7. Pump outlet temperature (1.0 and 1.5 pumps), Degrees Fahrenheit
    8. Cold box temperature, Degrees Fahrenheit
    9. Torque, lb-in
    10. Speed, rpm
- After 30 seconds of running, the ball valve was opened to simulate driver requested hydraulic function.
- Each test lasted for a total of 10 minutes, at which time the drive was turned off and data acquisition was terminated.
- Temperature of the cold box was re-adjusted for the next condition.
- Data was converted to Excel spreadsheet and plotted, emphasizing reduced flow rates as temperature drops.

At the completion of each lubricant temperature series, new oils were introduced into the system according to "Oil Drain, Flush and Fill Procedures for Cold Lubricant Pumpability Tests" document.

APPENDIX B
OIL DRAIN, FLUSH AND FILL PROCEDURES FOR COLD LUBRICANT PUMPABILITY TESTS

Oil Drain, Flush and Fill Procedures for
Cold Lubricant Pumpability Tests
Alan Montemayor
September 19, 2008

Drain Procedure:

1. Run the drive system and monitor reservoir temp till oil temperature reaches at least 30 °F.

2. Drain the lubricant into the original container(s), labeling the containers with a sticker that says, "Lubricant Name, Drain Date: xxxxxxx"

3. Take and label a quart sample with "Lubricant Name, Drain Date: xxxxxxx"

4. Break each line free and allow it to drain back to reservoir or to a container.

5. Finish draining the reservoir.

Flush Procedure:

1. Introduce enough of the next oil into the reservoir to cover both screens fully. This is about 7 gallons.

2. Run the system for five minutes to circulate the flush oil.

3. Drain the flush oil using the Drain Procedure except dispose of the flush oil.

Fill Procedure:

1. Fill the system with the next test oil till the fluid is at the top of the sight gauge.

2. Run the system to fill all lines. Monitor flow and pressure to assure pumping.

3. Drain oil to achieve the fill line (mid sight gauge).